MODERN VORTEX AND CENTRIFUGAL-VORTEX PUMP CONSTRUCTION

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Translation of "Sovremennyye konstruktsii vikhrevykh i tsentrobezhno-vikhrevykh nasosov."
Vestnik Mashinostroyeniya, No. 2, pp. 10-16, 1958.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
WASHINGTON   JULY 1965

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ABSTRACT

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The performance characteristics and constructional details of Soviet-made vortex and multistage centrifugal-vortex pumps, classified as open- and closed-passage types, are described, with numerous examples of specific commercial and experimental models. Some of the means incorporated for reducing the size and weight of such pumps, decreasing cavitation, economically introducing self-priming, and for enhancing pump efficiency are discussed, including the use of shrouds, series and parallel connection between stages, cantilever-type impellers, curvature of the impeller blades, configuration of the flow-through channel, use of a single impeller to serve different stages, multiple suction and discharge openings, etc. *Author*

Vortex pumps are widely used in various branches of industry, both in and /10* out of the Soviet Union. The theory and analytical design principles, as well as their areas of application in the national economy have been discussed previously (ref. 1). The present report simply describes the construction, design characteristics, and operation of modern vortex and centrifugal-vortex pumps.

The vortex pumps used most extensively outside the Soviet Union are of the open type, very similar in constructional detail to the Soviet commercial SVN-80 and VS-65-A vortex pumps, as well as vortex pumps of the closed type, with a peripheral flow passage, constructionally similar to the Soviet type V vortex pumps (ref. 2).

*Numbers in the margin indicate pagination in the original foreign text.

Prior to the 1950's, Soviet industry also produced these two types of vortex pumps almost exclusively, mainly the closed type. The principal advantages of the closed type of vortex pump over the open type are the steep Q-H curve of the former, higher efficiency and durability of the impeller blading, and somewhat higher engineering economy in their construction.

The drawbacks of this type of pump are their poor cavitation characteristics, the lack of a self-priming capability without added accessories, and low shaft r.p.m.

A common disadvantage of these pumps is the occurrence of relatively high radial thrust, which affects the working impeller. The radial thrust is given by

$$P = \frac{\gamma H D_2 B}{2},$$

where γ is the specific weight of the fluid, H is the pump head, D_2 is the outside diameter of the impeller, B is the width of the impeller.

Inasmuch as vortex pumps are used predominatly for large heads H, the radial thrust P can amount to a hundred kilograms, which tends to bend the shaft, cause wear on the impeller and casing, and prematurely wears out the bearings and stuffing box (ref. 3).

The first step toward elimination of the poor cavitation resistance of closed pumps was the design of the centrifugal-vortex pump, in which the first, auxiliary centrifugal stage generates the head necessary for cavitation-free operation of the second, or vortex stage.

As an example, we will consider the cavitation characteristics of the TsVS-53 pump (fig. 1).
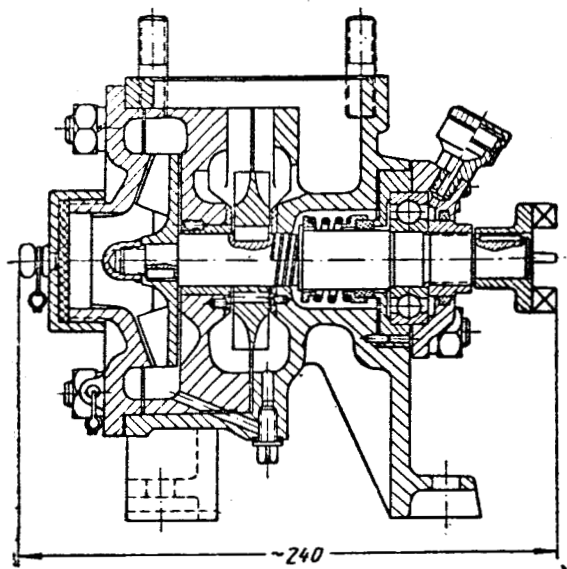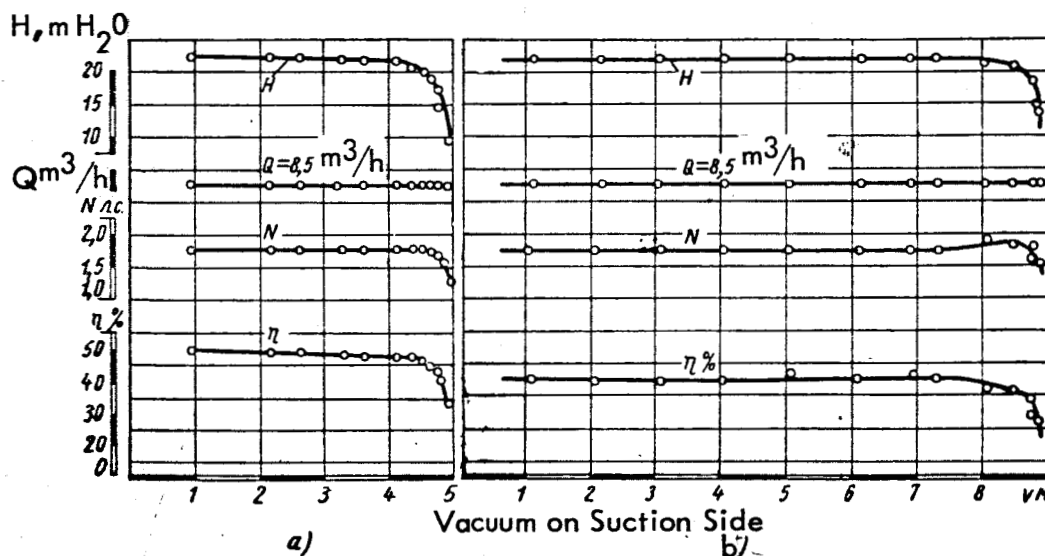
Figure 1. TsVS-53 Centrifugal-Vortex Pump.

Figure 2. Cavitation Characteristic of the TsVS-53 Pump and its Vortex Stage:

a) Vortex Stage Characteristic; b) Pump Characteristic.

Tests have shown that the closed type of vortex pump without the head

produced by the centrifugal impeller has a low cavitation resistance, which

almost cuts in half the permissible intake when fluids with a high vapor pressure are pumped under industrial conditions (fig. 2).

The placement of a centrifugal stage, with its high cavitation resistance, ahead of the vortex stage makes it possible to build closed type centrifugal-vortex pumps with as high as 6000 shaft r.p.m. This greatly reduces the weight and physical size of the pump, and improves its self-priming capabilities substantially. The size and weight of centrifugal-vortex pumps can also be diminished by combining the centrifugal and vortex impellers into one.

Pumps of the centrifugal-vortex type (type TsV) with a combined impeller are currently being produced on a regular basis. A typical example of the application of the old closed type of hydraulic section is the 2.5TsV-5 pump (fig. 3), which functions as part of the BMP-80M motopump system. Since the pump in this case is driven by a gasoline engine, one of the essential requirements is a maximally sloping head (and hence power) characteristic. This kind of characteristic has been successfully acquired by using inclined blades, curved back at an angle of 35°, on the vortex impeller.

In addition to their good hydraulic features, centrifugal-vortex pumps generally ought to have a high self-priming capability. In commercial operation, the self-priming rate is generally understood to be the time required for the fluid to rise in the suction pipe from its free level in the reservoir to its exit from the discharge pipe of the pump; this time depends on the capacity of the suction pipe and intake height. The automatic intake of fluid by centrifugal-vortex pumps is normally realized by means of a separating shroud placed over the discharge opening; this shroud has large weight and dimensions, and lowers the efficiency of the pump 8 to 12%.
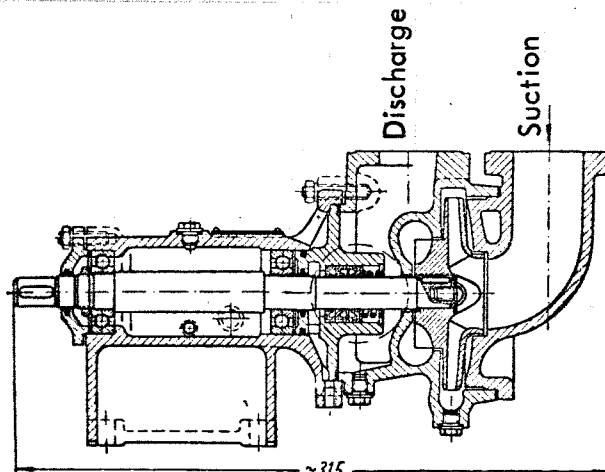
Figure 3.   2.5TsV-5 Pump with a Single Centrifugal-Vortex Impeller.

The self-priming capability of any pump equipped with a separating shroud is determined by the following factors: the nature of the pumped fluid (its vapor pressure, density, temperature, viscosity, and gas content); the length and diameter of the suction conduit; the geometric intake height; the shaft r.p.m.; the construction, shape, and dimensions of the separating shroud; the geometry of the flow-through part of the pump.

Figure 4 shows a conceptual diagram of a centrifugal-vortex pump in which the self-priming process is realized by means of a separating shroud. This type of separating shroud construction is the one most widely used at present. The self-priming process occurs as follows.

The casing of the pump is initially filled with the liquid to be pumped (say, 2 liters), then as the impeller rotates the vortex motion of the fluid creates favorable conditions for mixing of the air delivered from the suction pipe with the fluid filling the pump. The air-fluid mixture is discharged through the discharge opening into the shroud, which has a fairly large volume. As a result of the difference in density between the air and fluid, the mixture is separated in the shroud. The air is lifted upward, emerging into the
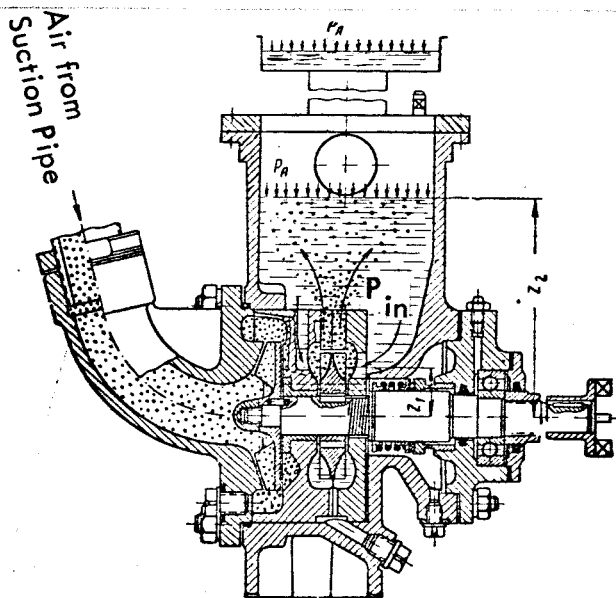
Figure 4. Type ESN-1 Pump with Separating Shroud.

discharge pipe, while the separated fluid is again delivered through a return

slot onto the blades of the impeller and again takes part in the mixture-forming

process. The separation process is continued until a quantity of air has been

removed from the suction conduit corresponding to the necessary pressure drop

for lifting the fluid.

When the fluid flow rate v_k into the return slot and the cross section F

of the latter are known, the flow rate of fluid through the return channel can

be determined, hence the required air suction intensity and dimensions of the

separating shroud can be properly calculated. The fluid flow through the return

channel is determined from the equation

$$Q = \mu F \sqrt{2g\left(z_2 - z_1 + \frac{P_{vac}}{\gamma}\right)}.$$

where μ is the discharge coefficient, $\mu = 1/\sqrt{1 + \xi}$ (where ξ is the friction

coefficient), z_1 and z_2 are the fluid levels in the pump, P_{vac} is the pressure

drop created by the pump in the suction pipe.

Investigations of shroud constructions have shown that increasing the dimensions of the shroud and fluid level in it (z_2) increases the suction intensity and the vacuum created, while decreasing the dimensions of the shroud and z_2 lowers the suction intensity. However, by placing dampers of various constructions over the delivery aperture and choosing the optimum fluid flow rate through the return slot, the air suction intensity can be greatly enhanced without increasing the dimensions of the shroud.

For pumps operating on fluids of different viscosities, the placement of dampers in the shrouds is mandatory, since strong liquid-air mixtures are formed in the case of a viscous fluid, and these mixtures are poorly separated in the separating shroud without dampers.

The use of grid dampers in the shroud is even more desirable in that, with their larger flow-through cross sectional area by comparison with aspirators, the pump efficiency is lowered only by 2 or 3%, whereas existing aspirator constructions lower the efficiency by 8 to 12%.

The effectiveness of grid dampers is confirmed by comparing the ESN-1 and TsVS-53 type pumps. Here, solely as the result of a structural modification of the shroud used in the ESN-1 pump and the addition of a grid damper, the TsVS-53 is created, which is only half the size and weight. The weight of the ESN-1 pump is 42 kg, that of the TsVS-53 is 18 kg, the efficiency of the TsVS-53 is 8% higher than that of the ESN-1.

In normal operation of the vortex pump with high heads, it is necessary to relieve the impeller of radial thrust.

One solution to this problem was the design of the STsL-16-75v pump by the Krasnyy Fakel Factory; this type of pump is illustrated in figure 5. By introducing a relieving crown in the upper part of the vortex stage, they were able
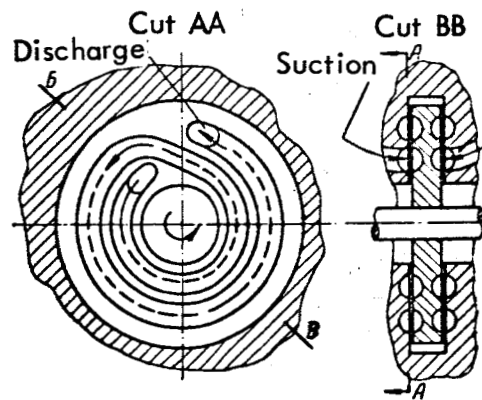
Figure 5. Diagram of the STsL-16-75v Pump.

to produce a very high head on one impeller. For this, two concentric rows of blades are placed on the impeller of the STsL-16-75v at different diameters, corresponding to the two working channels in the casing. The stages operate sequentially, increasing the head to 120 m in actual operation.

The absence of radial loading at such high heads meant that an experimental prototype of the pump could be tested on sliding bearings in water without additional lubrication, the wear turning out to be negligible.

It must be assumed that the indicated construction is the preferable one at the present time for generating high heads. However, it has the appreciable disadvantages of engineering complexity in production of the impeller and the difficulty of designing the self-priming equipment.

An alternate solution to the problem of relieving the impeller of radial thrust was the design of cantilever blades for the type 1.5V-1.3M impeller. Here, because the blades open both up and down, the pressure difference is absorbed by the pump casing, and the impeller remains unloaded. Furthermore, it becomes possible to impinge the fluid on the inner tips of the blades, which, first of all, permits additional utilization of the head from the action of the

centrifugal forces, second, somewhat improves the cavitation resistance of the pump (since the fluid is delivered at a smaller radius), and, third, constructionally facilitates the passage of fluid from one stage to the next for multistage pumps. Moreover, the milling or casting of the cantilever blades is slightly easier than for blades of the conventional type of impeller. However, cantilever blades require additional lathework and have somewhat less strength in bending.

For normal pump operation (without friction at the casing wall), however, these blades, like any other type of blade, have a considerable reserve of strength. Any blade will be put out of commission, of course, by stalling or choking.

The unquestionable advantages of cantilever blading (high head and high performance), which first became manifest in tests on the 1.5V-1.3M pump, have made it possible to design a number of centrifugal-vortex pumps with a single impeller relieved of any radial thrust.

This type of impeller was first used in the 3STsV-3 pump (fig. 6), which was designed to replace the double-impeller centrifugal-vortex pump STsL-20-24. The elimination of radial loading accomplished in the STsL-20-24 pump operating under 200 kg load made it possible to decrease the shaft diameter, to improve the seal, and to decrease the cost of operating the pump while lowering its weight and decreasing its size. In addition, by imparting a special configuration to the channel, the slope of the head characteristic under nonoperational conditions was rendered safe against engine overloading or destruction of the pipelines as $Q \rightarrow 0$. The effieiency under operating conditions is also increased (figs. 7 and 8). This type of pump has been further developed in the 1STsV-1.5, VS-2, 2STsV-0.8 and other pumps.
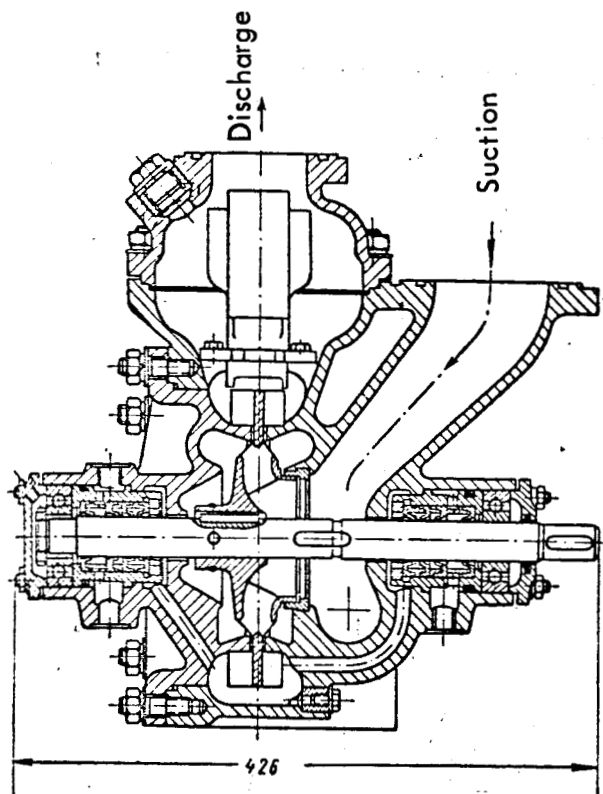
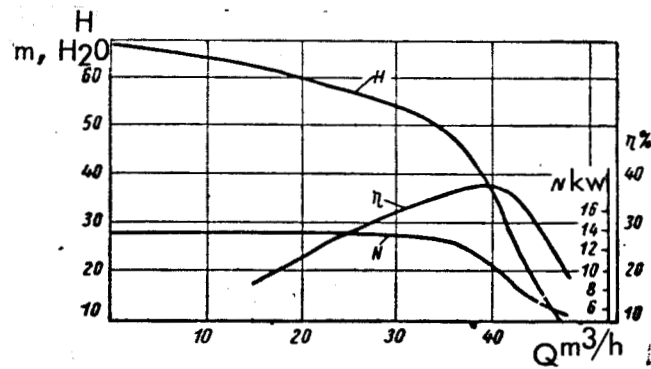Figure 6. 3STsV-3 Single-Impeller Centrifugal-Vortex Pump.

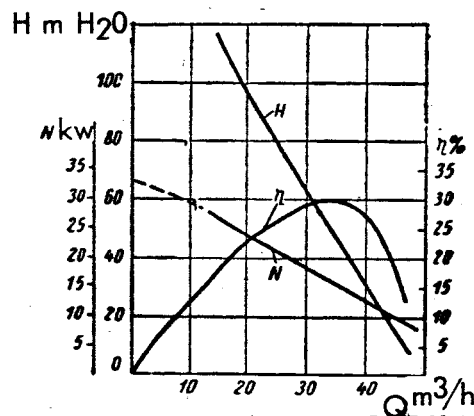Figure 7. Characteristic of the 3STsV-3 Pump.

Figure 8.   Characteristic of the STsL-20-24 Pump.

A highly representative model from the viewpoint of solving some of the most vexing problems in the construction of vortex pumps is the 2STsV-0.8 (fig. 9). Owing to the inclusion of a centrifugal stage, this pump operates at 3000 r.p.m. with sufficiently high cavitation resistance, the high r.p.m. imparting a good potential for the generation of high pressure on a single small impeller (D_2 = 175 mm). The pump impeller is relieved of radial thrust, thus permitting a head of 200 m to be obtained. The impeller in this case is seated on a long bracket on the shaft, making it possible to build the very strong stuffing assembly required at high pressures. Cantilever blades permit the hydraulically advantageous passage of fluid from the centrifugal to the vortex stage and on to the blade tips, as a result of which, in spite of the rather /14 severe parameters ($n_s \approx 8$; $Q = r$ m^3/h; H = 200 m H_2O; at a useful power of only about 2 kW the pump efficiency attains 36%).
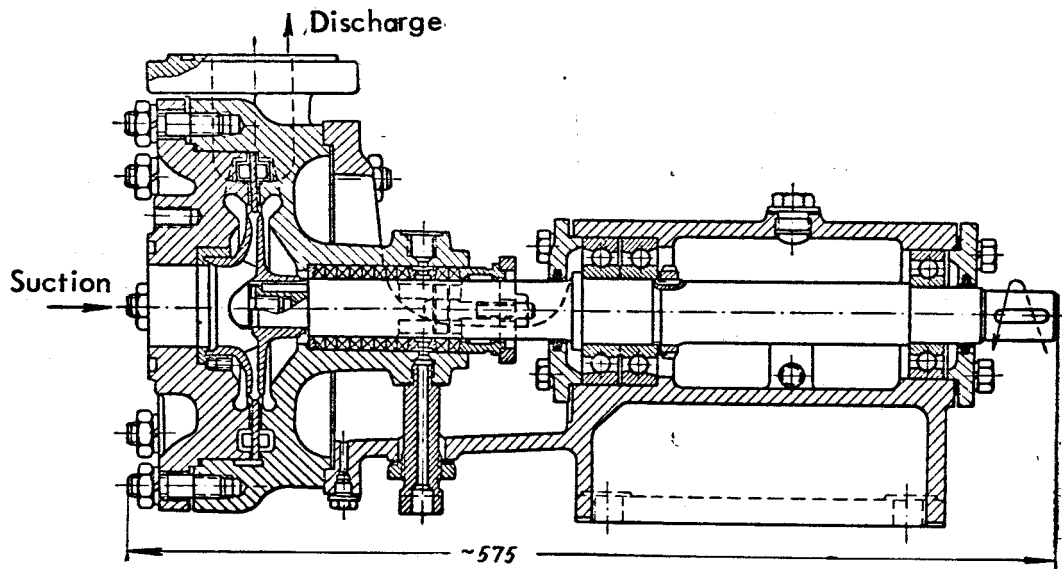
Figure 9. 2STsV-0.8 High-Head Pump with a Single
Centrifugal-Vortex Impeller.

In almost all vortex pumps, the liquid flow at a velocity v_1 in the discharge opening with cross section F_1, suddenly expanding from F_1 to F_2 as it emerges from the discharge opening and acquiring a velocity v_2, where $F_2 \gg F_1$. The discharge losses due to the sudden expansion of liquid flow on emerging from the discharge eye is determined by the Borda-Carnot equation:

$$h_{s.e.} = \frac{(v_1 - v_2)^2}{2g} .$$

The loss of head in sudden expansion can be expressed either in terms of the velocity head before expansion or in terms of the velocity head after expansion, i.e.,

$$h_{s.e.} = \left(1 - \frac{v_2}{v_1}\right)^2 \frac{v_1^2}{2g} = \left(\frac{v_1}{v_2} - 1\right)^2 \frac{v_2^2}{2g} .$$

Bearing in mind the equation of continuity $v_2/v_1 = F_1/F_2$, we obtain

$$h_{s.e.} = \left(1 - \frac{F_1}{F_2}\right)^2 \frac{v_1^2}{2g} = \left(\frac{F_2}{F_1} - 1\right)^2 \frac{v_2^2}{2g} .$$

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Consequently, decreasing the cross sectional area of the shroud is desirable not only for reducing the size and weight of the pump, it is also a very effective means of increasing the efficiency, because, for example, in the case of the ESN-1 pump, for which the cross sections of the discharge opening and shroud are $F_1 = 0.00027$ m^2 and $F_2 = 0.0256$ m^2 and the pump capacity is $Q = 10$ m^3/h, the head losses are

$$h_{s.e.} = \left(\frac{0.0256}{0.00027} - 1\right)^3 \left(\frac{10}{3600 \cdot 0.0256}\right)^2 \times$$
$$\times \frac{1}{2 \cdot 9.81} = 5.3 \text{ m}.$$

In the general balancing of the head developed by the pump, this amounts to about 15%. Consequently, an increase in the efficiency of vortex pumps must be realized both at the expense of an improvement in the configuration of the channel and impeller and at the expense of fluid flow on emerging from the discharge opening. The loss of head due to sudden expansion can be diminished as the result of carrying off the fluid from the discharge opening in helical fashion, using spiral volutes of the type common on centrifugal pumps; this permits elimination of the shroud.

This kind of volute was first realized in the construction of the VS-65-AM pump (fig. 10).

The cases that we have examined do not exhaust all of the possible ways of raising the efficiency or of decreasing the size and weight of closed type vortex pumps. The size and weight of the vortex pump can also be diminished by inserting a bracket with bearings "inside" the pump, as is done, for example, in the 2.5VS-3 pump.

This construction, while complicating the configuration of certain details only slightly, has made it possible to obtain the smallest possible size and weight for a given overall set of pump dimensions.
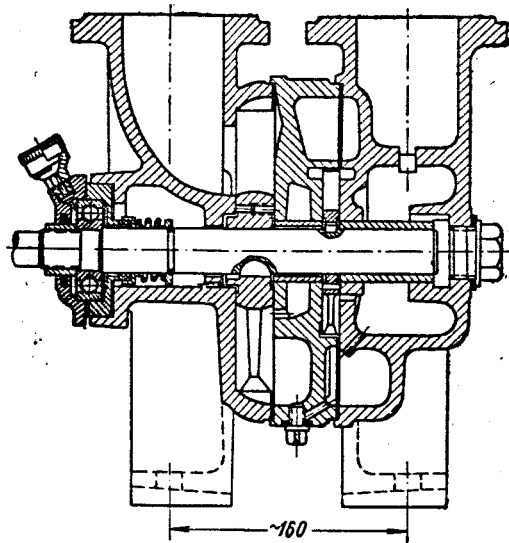
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Figure 10. VS-65-AM Vortex Pump.

A considerable shrinkage in size and weight, with improved operational

characteristics at the same time, has been achieved in new vortex pump construc-

tions with open passages. For example, the VS-65-AM pump, which has the same

self-priming capability and hydraulic characteristic in normal operating condi-

tions as the analogous VS-65A pump, has a three-percent higher efficiency. This

is attained chiefly through improvement of the flow-through part of the pump,

where the fluid from the working impeller is discharged by means of a rectangu-

lar spiral volute of the type used on centrifugal pumps. The axial dimension

of the new pump is half that of the VS-65-A, its weight is 35 kg instead of 62

kg. The construction and engineering are also far simpler.

Figure 11 shows a schematic diagram of the self-priming operation in cen-

trifugal pumps using open type vortex impellers. The vortex impeller in this

case is connected to the centrifugal impeller in parallel. The shortcoming of

such connection under actual pumping conditions is the possibility of fluid

backflow from the discharge pipe into the suction pipe, which greatly lowers the

capacity and performance of the pump. To avoid this, many factories have built
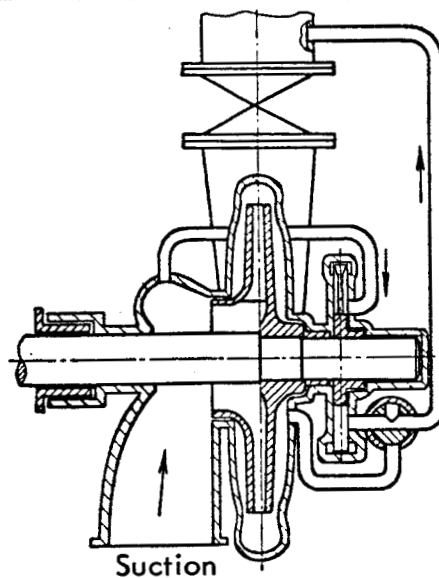
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Suction

Figure 11. Schematic Diagram of a Self-Priming Centrifugal Pump.

pumps designed such that the head characteristic of the vortex stage will be similar to that of the centrifugal stage. This also lowers the overall performance of the pump, however, due to the appreciable increase in power required by the vortex stage. The commercial TsSP-51 and L-PDM centrifugal pumps manufactured on this principle have an overall pump efficiency 8 to 12% lower than the efficiency of the principal centrifugal stage.

A more suitable hookup between the vortex and centrifugal stages is to connect the impellers in series, as is done in the new self-priming centrifugal pumps TsS-65 (fig. 12) and KSN-65. In these pumps, under self-priming conditions, the air passing through the centrifugal impeller and through the gap between the centrifugal impeller hub and casing is received in the chamber ahead of the vortex impeller, which pumps it into the discharge cavity of the vortex impeller. The air is then drawn off through a drainage pipe behind the discharge gate of the pump. Under actual pumping conditions, the vortex impeller is under the pressure developed by the centrifugal impeller, and the fluid is discharged
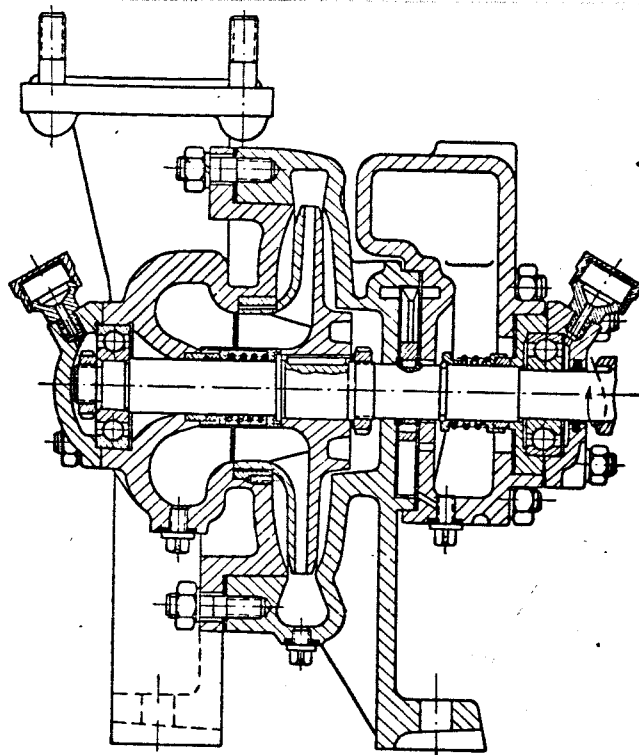
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Figure 12. TsS-65 Self-Priming Centrifugal Pump.

into the pressure well of the vortex stage with a head equal to the sum of the

heads of the centrifugal and vortex stages. This totally eliminates backflow

of the fluid, permitting the use of a vortex impeller with very small heads

(8 to 10 m H_2O), reducing the dimensions of the vortex impeller, and thus con-

siderably lowering the required power input; the efficiency of the centrifugal

stage is only lowered by about 4 or 5%.

Moreover, these constructions permit a twofold or more increase in the air /16

suction intensity of the usual vortex stage, since it becomes possible to space

two or more suction and discharge openings over the length of the channel

(fig. 13). The placement of the suction and discharge openings as in figure

13a provides the pump with self-priming, as well as a steeply falling Q-H curve.

With the openings placed as in figure 13b, the air suction intensity from the

suction line is increased and the head is lowered considerably during regular

pumping operation, which must lower still further the power required by the vortex impeller and raise the overall efficiency of the pump.
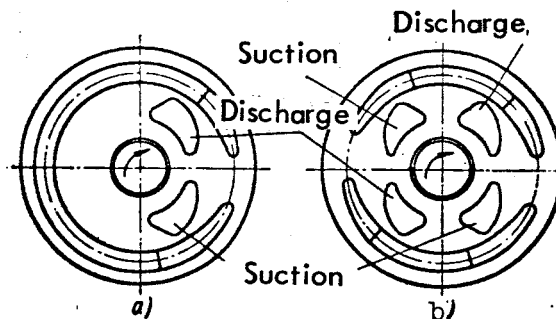

Figure 13. Placement of the Suction and Discharge Openings Along the Working Length of the Channel: a) With One Suction and One Discharge Opening; b) With Two Suction and Two Discharge Opening.

Considering the widespread application of vortex and centrifugal-vortex pumps in various branches of Soviet industry and the ever greater demand for them, it is essential that they be continually improved and refined, which will provide greater economy of ferrous and nonferrous metals, as well as improve substantially the hydraulic criteria and diminish the size and weight of such pumps.

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Translated for NASA by Stemar Engineering Inc.